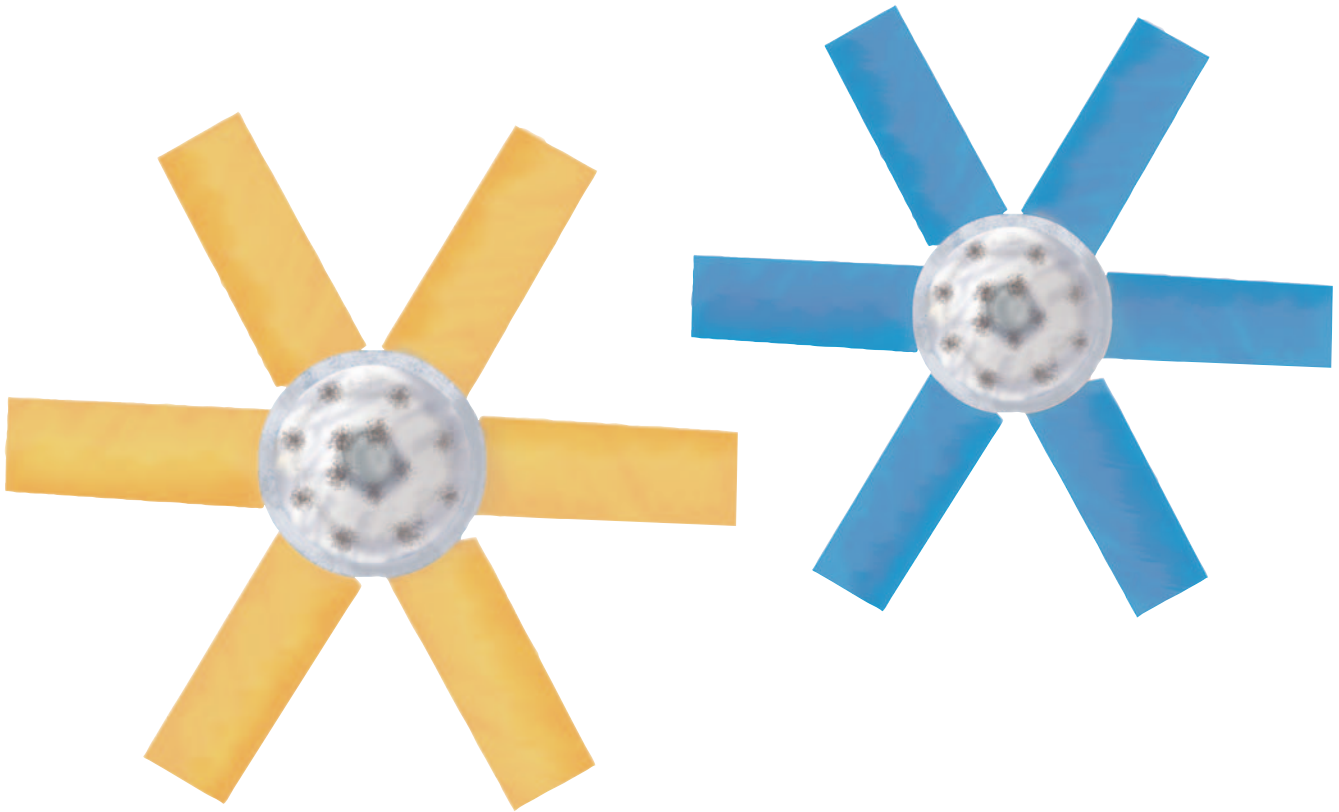




Fan Data

Is What You See, What You Get?

By **Kim G. Osborn**, Associate Member ASHRAE

It happens occasionally that even after taking great care in design, the air handler turns out to be louder than expected or the airflow is less than design. This article examines some discrepancies that can occur between catalog data and the actual performance.

At first it might seem strange that, with well-established testing standards, major differences can exist between the catalog data and the performance of the fan in application. A number of reasons exist for this.

Not every wheel size is tested, and those tested are run at a limited number of speeds. Fan testing is time consuming and

expensive. It is generally not practical to test all sizes or to test each size at a large number of speeds. Fan engineers strive to present the most reliable selection information they can, but due to gaps in the test series, not all sizes and operating points may be reliable. The end result is that you might take great pains to design an HVAC system with critical sound

and/or airflow issues, and substantially miss the design criteria because the fan data used was too optimistic.

It is becoming more important to be able to predict and control HVAC sound. Key to this goal is knowing the sound power spectrum of the fans used in the equipment. The designers then must either depend on the catalog data for the fans selected or measure the fans' sound power themselves. For designers without an available acoustical laboratory, the latter option is often too costly. Therefore, the only option is to use the catalog numbers.

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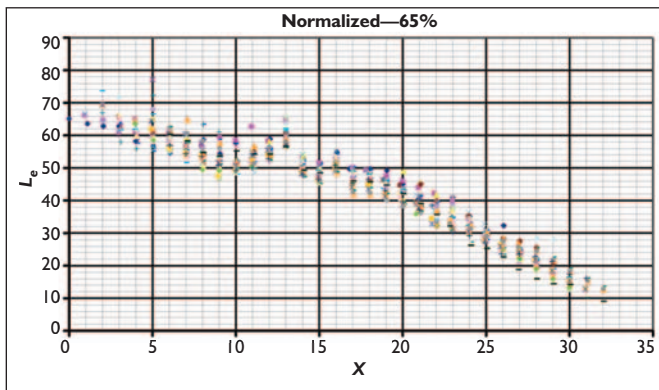


Figure 1: The discharge sound for a line of plenum fans.

What factors then affect the accuracy of the fan sound power catalog information? In the United States, most fan sound data is acquired under the Air Movement and Control Association International (AMCA) Standard 300-1996, *Reverberant Room Method for Sound Testing of Fans*. This lays out the details of performing reverberation room sound power testing of fans.

Since it would not be practical to test all sizes of fans at all possible operating points, there is a companion standard, AMCA Standard 301-1990, *Methods for Calculating Fan Sound Ratings from Laboratory Sound Data*, which defines methods to project test data to other wheel sizes and rotational speeds. The primary factors that control the quality of data, therefore, are the initial selection of test points (wheel sizes and rotational speeds) and how far these are projected in terms of wheel size and speed.

Ideally, one would test every wheel size of a model line at six or seven different speeds that totally bracket the usable range of each wheel size. This allows using interpolation instead of the AMCA Standard 301-1990 methods, and with careful selection of the test speeds, gives the most accurate sound data.

This proposal isn't necessarily practical. Besides involving a lot of testing, the upper end of the usable range of the larger fans exceeds the capacity of many laboratories.

Another consideration to fan manufacturers is that a fan line cataloged using this approach may not compare favorably to that of a competitor who does not sample in such detail.

Standard 301-1990 has two methods for calculating the sound of a fan line for all wheel sizes and at all speeds and operating points from the test data taken on a few samples. The preferred method is the "Generalized Sound Power Level Method" (Section 6.2), which projects sound power based on rpm and wheel diameter. Expressed simply, the test data is normalized, both in frequency and for wheel speed and diameter. Test data is plotted to a chart where the x-axis values are calculated from frequency and fan speed ($X = 10 \times \text{LOG}(\text{Frequency}/\text{rpm}) + 20$) and the y-axis values (L_e) are calculated from the measured sound power (L_w) with corrections for fan speed, wheel diameter, and band width ($L_e = L_w - 70 \times \text{LOG}(\text{Diameter}/20) - 10 \times \text{LOG}(\text{bandwidth}) - 40$

Frequency Ranges				
	Low	Blade Pass	High	Weighted Average
Operating points: 75%, 65%, and 55% of Wide Open Flow (WOF)				
rpm Coefficient	29	30	39	34
Diameter Coefficient	61	48	65	60
Operating points: 100%, 85%, 75%, 65%, 55%, and 50% of WOF				
rpm Coefficient	31	29	46	38
Diameter Coefficient	66	47	69	63

Table 1: Frequency coefficients calculated for the fan line.

24,723 cfm at 4.07 in. w.c. (11,669 L/s at 1,013 Pa) rpm = 1,250									
Measured	63	125	250	500	1 K	2 K	4 K	8 K	LwA
Discharge L_w :	96	94	102	100	96	92	90	82	102
Inlet L_w :	93	93	97	87	88	86	83	77	94
Catalog Values at 1,151 rpm (same airflow)									
Discharge L_w :	88	92	99	94	90	84	81	77	96
Inlet L_w :	85	91	98	85	82	76	73	71	91
Deviation									
Diff. Discharge:	8	2	3	6	6	8	9	5	6
Diff. Inlet:	7	2	-1	2	6	10	10	6	3

Table 2: Measured sound of test wheel compared to catalog data.

$\times \text{LOG}(\text{rpm}/1000)$). These plots are made for both the discharge and inlet sound for each normalized point of operation. These points are usually expressed as a percentage of the airflow at zero static pressure (wide open flow). To project the sound power at a different speed and/or wheel diameter, the x-values for each frequency are calculated from the new rpm and, using the curve for the appropriate operating point, the y-values are read from the curve. These values then are corrected for the new wheel speed, wheel diameter, and band widths. Curves can be interpolated between tested points of operation.

In a perfect world, all test data for the same relative point of operation for all speed and wheel sizes of a fan model would plot onto one, well-defined curve. Figure 1 illustrates that this is not a perfect world. The data presented is the discharge sound for a line of plenum fans from 18.25 in. (464 mm) to 60 in. (1524 mm) operating at 65% of the wide open flow point. Each wheel was tested at two or three speeds. The test speeds were selected to place the blade pass tone at the center of a third octave band. These fans were random production models equipped with motors, belts, and sheaves. They were not optimized for catalog testing.

It is easy to see that if the project's economics only allow the testing of three or four wheel sizes, and these at only one speed each, the data collected might not be representative of the full line at all speeds.

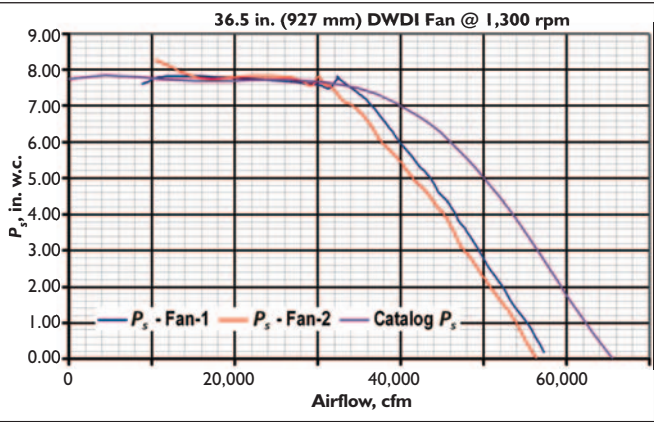


Figure 2: Measured fan airflow data compared to catalog data.

One way of improving the situation is to modify the implementation of the standard a little by regressing the best rpm and diameter coefficients for a given fan line instead of using the standard coefficients of 40 and 70. For this fan model at least, this leads to another problem. What data should be used in the calculations? Should you concentrate on the operating points most often used, say between 75% of wide open flow and 55%, or include all test points from the wide open flow point to close to the stall point? From these test sets, what frequencies should be included?

The fan spectrum can be divided into three regions. The third

43,679 cfm at 5.48 in. w.c. (20,616 L/s at 1,363 Pa)										rpm = 1,333
Measured: Fan-1	63	125	250	500	1 K	2 K	4 K	8 K	LwA	
Ducted Discharge L_w :	104	99	105	101	96	94	86	79	103	
Inlet - L_w :	104	100	107	96	96	95	86	79	102	
Measured: Fan-2										
43,381 cfm at 5.01 in. w.c. (20,476 L/s at 1,246 Pa)										rpm = 1,333
Ducted Discharge L_w :	105	99	102	99	95	91	87	82	101	
Inlet - L_w :	100	100	107	95	94	91	86	82	101	
Projected from Fan Program (at 1,238 rpm)										
Discharge/Inlet L_w :	110	110	107	100	97	93	84	75	103	

Table 3: Catalog sound data for tested axial fans.

octaves below the blade pass frequency, the third octave including the blade pass frequency, and the third octave bands above the blade pass. Each zone converges best with different coefficients. Table 1 shows the coefficients calculated for this fan line. The Weighted Average column is the average of the three values with the high frequency coefficient counted twice, as this is the largest area.

As you can see, the standard values of 40 and 70 are not that far off for most of the spectrum (those above the blade pass frequency). However, they do not do as well projecting the blade pass tone or the bands below the blade pass, which often contain tones at harmonics of the rotational frequency. Prob-

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ably the best solution here is to increase the data density. That is, test more sizes at a wider range of speed to reduce the difference between the design point and tested points.

In some fan lines, there are physical changes in the wheels as the wheel sizes or service class changes. If these changes affect the airflow and sound and are not adequately sampled in the test program, then the sound and airflow will not be projected properly for all wheel diameters or operating points. For instance, different wheel sizes for some fan models have different numbers of impellers (blades), which means that if the test wheels are not carefully selected, the blade pass tones will not be properly projected for some wheel sizes.

Sometimes the construction of the wheels change substantially with class. Some manufactures will place a mid-blade stiffening rib in the class II wheels which are not present in the Class I wheels. We recently tested such a class II wheel. Compared to cataloged data, airflow of the test wheel was approximately 8% low. *Table 2* shows a comparison of the measured sound to the catalog data.

Since the fan is marketed as a new, significantly quieter design, the results were a little disappointing. Upon investigation, however, it was determined that the catalog data was generated from a Class I wheel. I understand this situation is being remedied.

Although most of the testing presented here has been with plenum fans, similar effects occur with other types of fans. Included here are two examples. One example is for a double width double inlet (DWDI) centrifugal fan, and the other is for an axial fan.

The results of a recent testing program have indicated a larger than expected variance from the catalog data for some DWDI models. Some of this may be the results of only using 2 to 2.5 equivalent diameters of duct rather than the recommended 4 to 5. These tests primarily were designed to acquire sound data. Two diameters are long enough to generate an end reflection to be corrected (AMCA Standard 300-1996) and simplify the test setup. Our experience is that

it is rare to find 25 ft (7.6 m) of straight duct on a fan in real-world applications. Most of the differences are probably the result of unidentified system effects and what is sometimes called manufacturing variances.

The most dramatic difference between the test data and catalog data is shown in *Figure 2*. This shows the measured airflow

data for two nearly identical 36.5 in. (927 mm) DWDI fans and compares these to the catalog data. The two test fans used the same size and design of wheels, inlet cones, and scrolls but had minor differences in the support frame design. Not only did these two “identical” fans have noticeably different airflow, but both fell substantially short of the cataloged airflow curve.

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We are still looking into the causes behind these variations, but the most probable reason is that this particular fan design is relatively sensitive to small manufacturing variations and both of these fans are probably near the edge of allowable variances.

As mentioned, these fans also were being tested for sound. The results have been somewhat mixed, but for the most part, the 36 in. (927 mm) fan data was typical of the others tested. The catalog data for these fans is actually pessimistic at the low end of the spectrum but a little optimistic at the high end (*Table 3*).

The final example comes from tests of axial fans. *Figure 3* shows the airflow test results for one of the three fans tested.

The Catalog-Ducted curve shows the catalog airflow for this fan. The first key here is the term “ducted.” Often, one has to look carefully at the data sheet to pick this up. This is a frequent cause of airflow problems, as this is often missed or ignored. As these fans were to be imbedded in the middle of the units and were not ducted, the “Catalog est. Un-Ducted” curve is this same curve adjusted for an un-ducted configuration.

Keep in mind the instructions for calculating the un-ducted corrections often are placed in an appendix or in a totally different publication rather than being shown on the data sheets, and not always built into the selection computer program.

The Original Wheel curve in *Figure 3* shows the tested

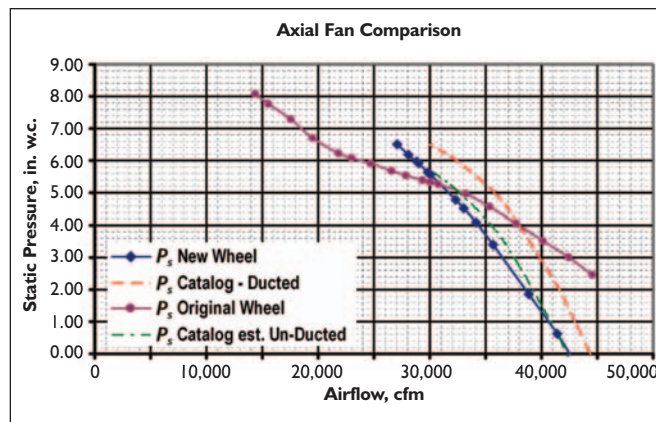


Figure 3: Airflow test results of one of three axial fans.

airflow data for one of the fans originally received from the manufacturer. If the design point had been near or to the right of the point where the curves cross, there wouldn’t have been a problem. Unfortunately, the design point had been close to the top end of the catalog curve.

We were especially confused by the fact that the measured curve had a substantially different slope than the catalog curve. The manufacturer believed that the problem was that we were testing these fans without ducts, and felt that axial fans were

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supposed to be tested ducted. They did send replacement wheels but they didn't have exact replacements available.

The wheels received consisted of a somewhat different hub from a different fan model combined with a design of impeller that was not being used anymore but of which they had a few boxes left over. The New Wheel data shows the airflow data for this configuration. Surprisingly, this combination matched the design airflow curve pretty well.

I mentioned that this would not have become an issue if the fans had not been selected at the peak of the curve. Actually, selecting at the upper end of any published curve for all types of fans is another frequent source of problems. The upper end, near the stall point of the fan, generally represents what the fan can achieve if everything comes together just right without pesky system effects. Besides the well-known cautions that selecting a fan near the stall point leaves you vulnerable to errors in the calculations of the building static pressures, the individual sample you received may stall at a lower static pressure than shown on the curve due to manufacturing variations or overlooked system effects. Of course, this throws the fan immediately into a stall condition even if the building static pressures were calculated correctly.

So where does this leave us? For many jobs, it is perfectly safe to use the catalog data. Just avoid selecting fans too close to the maximum static pressure shown on the curve and remember to select motors of sufficient power that the fans can be sped up some if the airflow is low or the stated power requirements are optimistic. For instance, if the catalog fan data suggests that the fan input power will be 29.5 hp (22.0 kW). Do not select a 30 hp (22.4 kW) motor as no room exists to increase the fan speed.

For jobs where sound is critical, the ideal solution is to test the fans yourself. If this is not practical, try to get information on how the fan line was tested. Specifically, find out which wheel sizes were tested and at what speed. This will tell you how far away your operating point is from actual test data and will at least set a level

of confidence. If the same size wheel was tested at close to the design rpm, then you should only need to allow for manufacturing variations. If the projection is a large one, that is several wheel sizes and/or a substantial change in rpm, allow a larger safety factor in your calculations.

Finally, if the engineering community shows enough concern about the reli-

ability of fan data, at least some manufacturers will see a business opportunity in increasing their testing efforts and marketing the concept that their data is more reliable. These manufacturers may then be willing to support or even promote the concept of open market testing by a certification agency to validate published data. ●

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